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Effectiveness of Rotary Air Preheater in a Thermal Power Plant

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Abstract: Energy saving is one of the key issues, not only from the viewpoint of fuel consumption but also for the protection of global environment. Large quantity of hot flue gases is released from boilers, kilns, ovens and furnaces. If some of this waste heat could be recovered, a considerable amount of primary fuel could be saved. One of the efforts has been made for heat recovery in Thermal Power Plants using rotary regenerators. The present study is on the performance analysis of rotary air preheater in a Thermal power plant. In this thesis, the performance of air preheater is evaluated based on the operating conditions such as mass flow rate and temperature of both air and flue gases. In this analysis the variation of Reynolds number, Colburn j factor and, Friction factor at the operating conditions are studied. The effectiveness of the air preheater is evaluated using Kays and London correlation and compared with the counter flow heat exchanger at the same operating conditions of air preheater. The variation of pressure drop, heat transfer coefficient, and heat transfer rate at different mass flow rate of air observed.

Keywords: Air preheater, Colburn j factor, Friction factor, Heat transfer coefficient, Pressure drop, Effectiveness.

I. INTRODUCTION

Rotary air preheater is the main equipment used to recover waste heat from the exhaust flue gases in thermal power plants. By preheating the combustion air with the hot flue gases leaving out of the boiler, a considerable increase in efficiency is obtained. The ducts of hot gas and cool air are arranged in such a way that both the flue gas and the inlet air flows simultaneously through the air preheater as shown in fig-1. The air preheater absorbs the heat energy from the flue gases and stores in the matrix bed. The air from the surroundings is allowed to flow through the matrix bed and gets heated up. The heated air from the air prehater flows through two ducts as primary air and secondary air. The primary air enters into the mills to dry the coal and transport the pulverized coal to burner and, the secondary air is used for better combustion in furnace.

Several researchers discussed performance analysis of air preheater used for different purposes and developed correlations both numerically and experimentally. Air preheater is one of the heat exchanger, Shanker and Kishore [1] in their paper carried out performance evaluation on charge air coolers with varying mass flow rates on hot side from and keeping cold fluid rate is constant, and found that water-cooled charge air cooler is having higher effectiveness than air-cooled type. Saunders and Smoleniec [2] suggested that the effectiveness of fixed bed regenerators for cases with equal reduced periods and lengths could be compared to cases with unequal reduced periods and lengths. Mounika et al. [3] in their paper studied about the performance analysis of automobile radiator. It is similar to cross flow heat exchanger which is designed to transfer the heat from the hot coolant coming from the engine to the air blown through it by the fan. A small segment of the radiator is analyzed for the various speed of the air striking the radiator as the vehicle moves from its rest position to a certain speed. Skiepko [4] in this paper is to compare results obtained based on theoretical modelling with directly measured experimental data on a full scale operating air preheater. London et al. [5] developed differential equations describing the transient behaviour of temperatures in rotary regenerators, obtaining analytical solutions supplemented by an experiment using an electrical analog system and numerical solutions. Bahnke and Howard [6] obtained numerical solutions for steady state temperatures in a rotary regenerator, including the effect of heat conduction in the matrix material along the flow direction. Rajnish kumar and Kishore [7] in this paper, an experimental study of the condensation of water vapor from a binary mixture of air and low-grade steam has been depicted. The study is based upon diffusion heat transfer in the presence of high concentration of non condensable gas. To simplify the study, experimental analysis is supported by empirical solutions. The main objective of this work is to establish an approximate value for surface area and overall heat transfer coefficient of a horizontal shell and tube condenser used in process space. Sepehr Sanaye and Hassan [8] in this paper the pressure drop and effectiveness of rotary regenerator are important parameters in optimal design of this equipment for industrial applications. For optimal design of such a system, it was thermally modelled using ε-Ntu method to estimate its pressure drop and effectiveness. Sreedhar Vulloju et al. [9] tested air preheater elements using cold flow studies. He proved that performance of Ljungstrom air preheater is dependent on the heat transfer element profiles. It is necessary to develop element profiles with lesser pressure drops for efficient heat transfer with lower power consumption to improve overall efficiency of power plant.

II. AIR PREHEATER

The function of air preheater is to increase the temperature of the air before it enters the furnace. It is generally placed after the economizer, so that the flue gases pass through the economizer and then to the air pre heater. The Ljungstrom

Air preheater is a try sector type as shown in fig-2 with rotor diameter and hub diameter are 8.89m, 1.42m respectively. The height of the heating elements of three sections are respectively 450mm, 1050mm, and 300mm from top to bottom of the rotor. The cold end heating elements are made of Corten steel while the hot and intermediate end heating elements are made of Carbon steel. The heating elements present in all three sections are double undulated element profiles having hydraulic diameter (D_h) is 0.7475 mm. The total mass of the air preheater is approximately 144 tons. Total surface area of the air preheater or total matrix surface area is 17000 m^2, and the matrix bed porosity is 0.78.

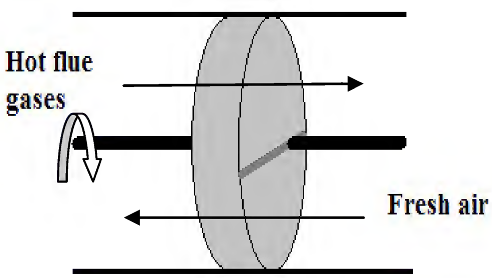

Fig-1: Flow passages in the rotary regenerator

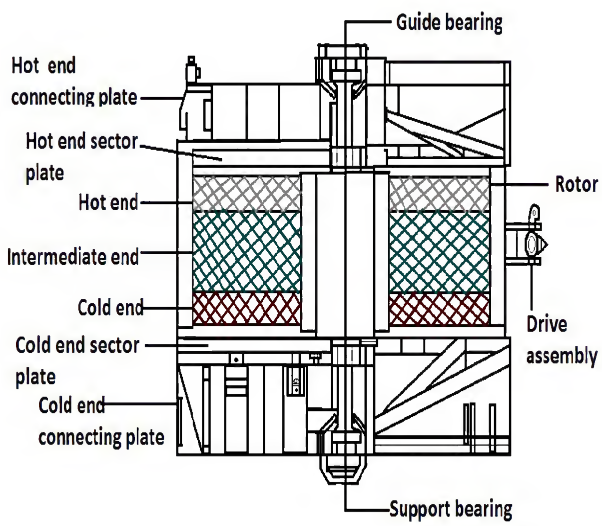

Fig-2: Air preheater

The rotary air preheater is a counter flow regenerative rotary type heat exchanger. Specially corrugated heating elements are tightly placed in the sector compartment of the rotor. The rotor turns at a speed of 1.42 rpm and is divided into gas channels and air channels. The air side is made of primary air channels and secondary air channels. When gas flows through the rotor, it releases heat and delivers it to the heating elements and then the gas temperature drops, when the heated elements turn to the air side, the air passing through them is heated and its temperature is increased. By continuing maintaining such a circulation, the heat exchange is achieved between gas and air.

Usually air leaks in to the gas in the air preheater due to pressure differences. This leakage air decreases the flue gas temperature without extracting the heat. To reduce the air leakage seals are provided. It is an implied requirement that the rotating parts should have some working clearance between the static parts to avoid any interference between them. Here, in air preheaters, rotors are constructed to have high thermal expansion and these gaps are close with the flexible seal leaves. Major types of seals used in thermal power plant are radial seals, axial seals, bypass seals, and circumferential seals.

III. DESIGN THEORY OF AIR PREHEATER

The differential equations are subject to the normal idealizations made in the design theory. The differential equations can be made dimensionless to arrive at the dimensionless groups for the effectiveness - number of transfer units (ε-Ntu_o) or the reduced length - reduced period (Λ-Π) methods of solution. The ε-Ntu_o method is usually used for rotary regenerator. It can be shown that the effectiveness of a counter flow regenerator is a function of six dimensionless variables.

$$\varepsilon = f\,(Ntu_{o,}\ (hA)^*,\ C_r^*,\ C^*,\ \lambda,\ A_k^*)$$

The most important assumption in the above relation is that there is no pressure or carryover leakage. In a direct contact type recuperator the effectiveness is only a function of Ntu_o and C*. The parameters $(hA)^*$ and C_r* arise because of the heat storing nature of a regenerator. The longitudinal heat conduction effect is accounted for by λ and A_k^*. An advantage of this particular choice of dimensionless parameters is that the influence of $(hA)^*$ and A_k^* on the effectiveness are small and that the solution method parallels that of a recuperator for the case of infinite C_r^* (infinite rotational speed), so the effectiveness as a function of the six above-mentioned parameters. In the case of zero longitudinal heat conduction the functional relationship for the effectiveness becomes

$$\varepsilon = f\,(Ntu_{o,}\ (hA)^*,\ C_r^*,\ C^*)$$

As mentioned earlier, the wall temperature profile in a regenerator (in the absence of longitudinal wall heat conduction) is going to be dependent on the thermal conductance's $(hA)_h$ and $(hA)_c$ between the matrix wall and the hot or cold fluids. For a high temperature regenerator, the thermal conductance will not only include convection conductance but also radiation conductance. The dimensionless group that takes into account the effect of the convection conductance ratio is $(hA)^*$. Lambertson (1958) and others have shown through a detailed analysis that $(hA)^*$ has a negligible influence on the regenerator effectiveness for the range $0.25 \le (hA)^* < 4$. Since most regenerators operate in this range of $(hA)^*$, fortunately, the effect of $(hA)^*$ on the regenerator effectiveness can usually be ignored.

Convection conductance ratio $(hA)^* = \frac{(hA) \text{ on the } C_{min} \text{ side}}{(hA) \text{ on the } C_{max\,side}}$

Now the effectiveness of a counter flow regenerator is a function of three dimensionless variables.

$$\varepsilon = f\,(Ntu_o,\ C_r^*,\ C^*)$$

For specified Ntu_o, C^*, and Cr^*, the effectiveness generally decreases with decreasing values of $(hA)^*$, and the reverse occurs for large values of Ntu_o and $C^* \approx 1$. However, the

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influence of $(hA)^*$ on ε is negligibly small for $0.25 \le (hA)^* < 4$, as shown by Lambertson, among others. When C_r^* tends to ∞, the effectiveness ε of a regenerator approaches that of a recuperator. The difference in ε for $C_r^* \ge 5$ and that for $C_r^* = \infty$ is negligibly small and may be ignored for the design purpose.

IV. ANALYSIS OF AIR PREHEATER

The performance of air preheater can be analyzed by putting the necessary equations in order as below:

Frontal or face area of air preheater is given by

A_{fr} = Rotor cross sectional area × (fraction of rotor face area not covered by radial seals) (1)

Rotor cross sectional area = disk area(πR^2) – hub area(πr^2)

The flue gas and air side frontal areas are proportional to their respective flow split ratio. The flow split ratio of air preheater is 1:1

Now hot side frontal area, and cold side frontal is given by

$$A_{frh} = \left(\frac{A_{fr}}{2} \times \sigma\right) \qquad A_{frc} = \left(\frac{A_{fr}}{2} \times \sigma\right) \tag{2}$$

Mass velocity of flue gases is given by

$$G_f = \frac{m_f}{A_{frh}} \tag{3}$$

Reynolds number of flue gases is determined by

$$Re_f = \frac{G_f D_h}{\mu_f} \tag{4}$$

Colburn j factor and Friction factor data for the matrix surfaces in rotary regenerators are presented as experimental correlations in terms of Reynolds number. These correlations for some matrix surfaces are given by Kays and London. The rotor of this air heater is enclosed in a causing, the cross section of it can be square, triangle, or orthogonal. For large diameter rotor the orthogonal casing is provided.

Colburn j factor of flue gases (j_f) is given by

$$j_f = \exp[\, E + F (\ln Re_f) + G (\ln Re_f)^2 + H (\ln Re_f)^3 \,] \tag{5}$$

Friction factor of flue gases (f_f) calculated from the following correlation

$$f_f = \exp[\, A + B (\ln Re_f) + C (\ln Re_f)^2 + D (\ln Re_f)^3 \,] \tag{6}$$

Stanton number of flue gases (St_f) is given by

$$St_f = (\, j_f \times Pr_f^{-2/3}) \tag{7}$$

The heat transfer coefficient of flue gases (h_f) can be calculated from the following correlation

$$h_f = (St_f \times G_f \times Cp_f\,) \tag{8}$$

Similarly the above calculations can be done to air side also.

Modified number of transfer units is given by

$$Ntu_o = \frac{1}{C_{min}\left[\frac{1}{(hA)_f} + \frac{1}{(hA)_a}\right]} \tag{9}$$

Surface area of the both streams are proportional to the respective flow split ratio, so that

$$A_f = \frac{A}{2}, \text{ and } A_a = \frac{A}{2}$$

Now overall heat transfer coefficient (U_o) is given by

$$\frac{1}{U_o A} = \frac{1}{(hA)_f} + \frac{1}{(hA)_a} \tag{10}$$

Now let us present approximate formulas to compute effectiveness (ε) for a wide range of Cr* and C*. The influence of Cr* on effectiveness (ε) can be presented by an empirical correlation for ε ≤ 90% by Kays and London.

Now finally the effectiveness of air preheater is given by

$$\varepsilon = \varepsilon_{cf}\left[1 - \frac{1}{9(C_r^*)^{1.93}}\right] \tag{11}$$

Where

$$\varepsilon_{cf} = \frac{1 - e^{[-Ntu_o(1-C^*)]}}{1 - C^* e^{[-Ntu_o(1-C^*)]}} \tag{12}$$

$C_r = (M \times C_m \times N)$

Heat transfer rate (Q) from the flue gas stream to the air stream is given by

$$\varepsilon = \frac{\text{actual heat transfer rate}}{\text{maximum possiable heat transfer rate}}$$

$$\varepsilon = \frac{Q}{Q_{max}}$$

$$Q = \varepsilon \times Q_{max} \tag{13}$$

The expression for evaluating pressure drop by kays & London correlation

Pressure drop in the flue gas flow is given by

$$\Delta p_f = \frac{G_f^2}{2}\left[4 f_f \frac{L}{D_h}\left(\frac{1}{\rho_m}\right) + (1+\sigma^2)\left(\frac{1}{\rho_{fi}} - \frac{1}{\rho_{fo}}\right)\right] \tag{14}$$

Pressure drop in the air flow is given by

$$\Delta p_a = \frac{G_a^2}{2}\left[4 f_a \frac{L}{D_h}\left(\frac{1}{\rho_m}\right) + (1+\sigma^2)\left(\frac{1}{\rho_{ao}} - \frac{1}{\rho_{ai}}\right)\right] \tag{15}$$

Effectiveness (ε_o) of the Counter flow heat exchanger is given by

$$\varepsilon_o = \frac{m_a C_{pa}(T_{ao} - T_{ai})}{C_{min}(T_{fi} - T_{ai})} \quad \text{(or)} \quad \frac{m_f C_{pf}(T_{fi} - T_{fo})}{C_{min}(T_{fi} - T_{ai})} \tag{16}$$

V. RESULTS AND DISCUSSION

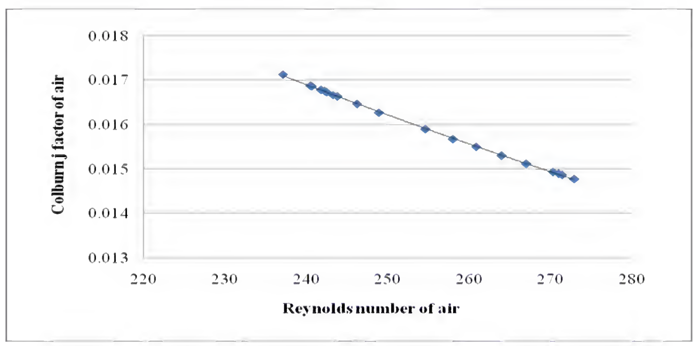

Fig-3: Variation of Colburn j factor of air Vs Reynolds number of air

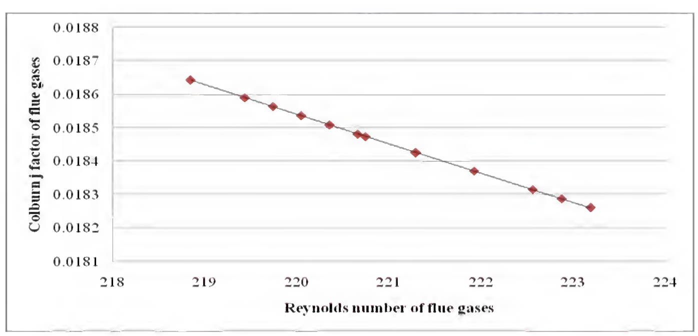

Fig-4: Variation of Colburn j factor of flue gases Vs Reynolds number of flue gases

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The variation of Colburn j factor with Reynolds number of both streams is shown in the Fig-3 and Fig-4. The result shows that the Colburn j factor decreases with increase in Reynolds number. Colburn j factor is an exponential function of Reynolds number. With increase in the mass flow rate of air entering into the air preheater the Reynolds number of air increases there by the Colburn j factor decreases. In this analysis mass flow rate of flue gases entering into the air preheater is same. The flue gases temperature decreases while passing through the air preheater, so viscosity of the of the flue gases decreases. Reynolds number of flue gases increases due to decrease in viscosity, so there is a decrease in Colburn j factor. The range of Reynolds number in this analysis is less, so the curve looks like approximate linear.

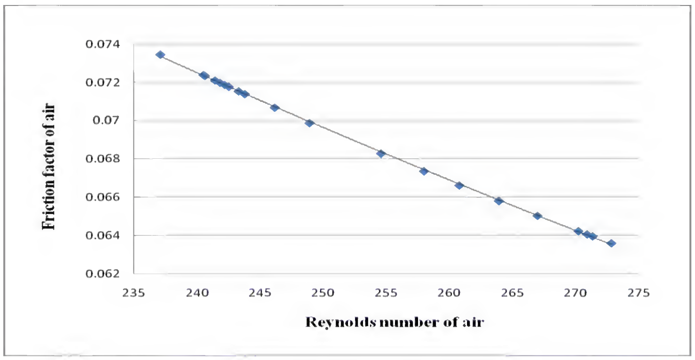

Fig-5: Variation of Friction factor of air Vs Reynolds number of air

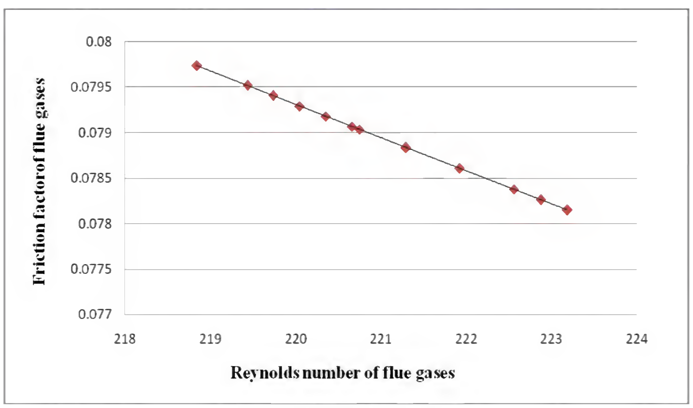

Fig-6: Variation of Friction factor of flue gases Vs Reynolds number of flue gases

The above Fig-5and Fig-6 shows the variation of Friction factor with Reynolds number of both streams. The result shows that the Friction factor decreases with increase in Reynolds number. Friction factor is an exponential function of Reynolds number. With increase in the mass flow rate of air entering into the air preheater the Reynolds number of air increases there by the Friction factor decreases. In this analysis mass flow rate of flue gases entering into the air preheater is same. The flue gases temperature decreases while passing through the air preheater, so viscosity of the of the flue gases decreases. Reynolds number of flue gases increases due to decrease in viscosity, so there is a decrease in Friction factor. The range of Reynolds number in this analysis is less, hence the curve looks like approximate linear.

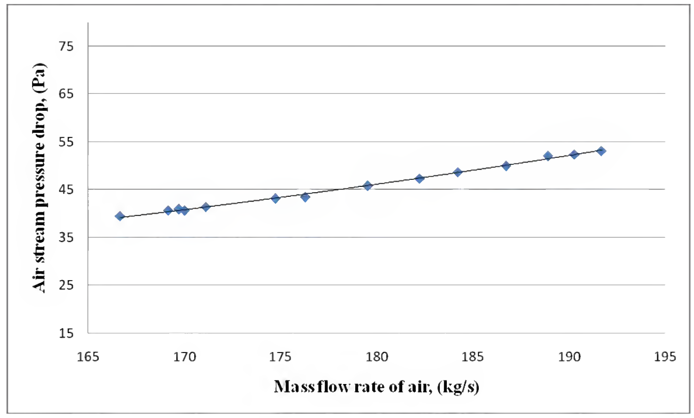

Fig-7: Variation of Air stream pressure drop (Pa) Vs Mass flow rate of air (kg/s)

The variation of air stream pressure drop (Δp) with mass flow rate of air (m_a) is shown in the Fig -7. The result shows that the air stream pressure drop increases with increase in mass flow rate of air. Pressure drop is a polynomial function of mass velocity (G) and Friction factor. As mass flow rate of air increases mass velocity increases, but there is decrease in Friction factor. The decrease in Friction factor is negligible when compared to increase in mass velocity. The range of mass flow rate in this analysis is less, so the curve looks like approximate linear.

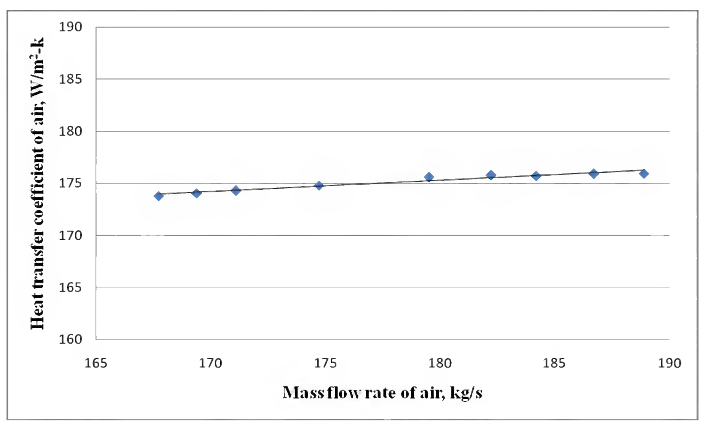

Fig-8: Variation of Heat transfer coefficient of air Vs Mass flow rate of air

The above Fig-8 shows the variation of heat transfer coefficient of air (h_a) with mass flow rate of air (m_a). The result shows that the heat transfer coefficient of air (h_a) increases with increase in mass flow rate of air. Mass velocity of air increases with increase in mass flow rate of air, hence the heat transfer coefficient of air increases.

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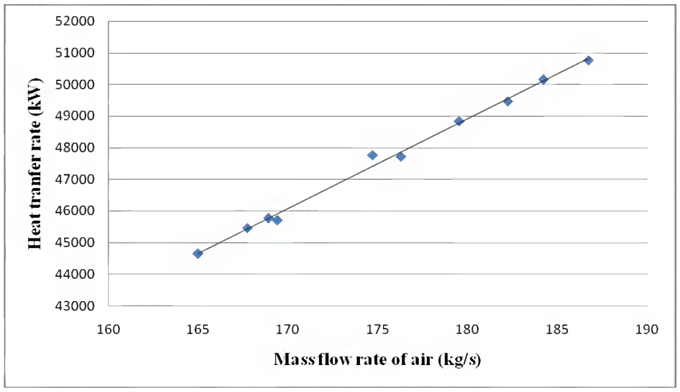

Fig-9 Variation of Heat transfer rate (kW) Vs Mass flow rate of air (kg/s)

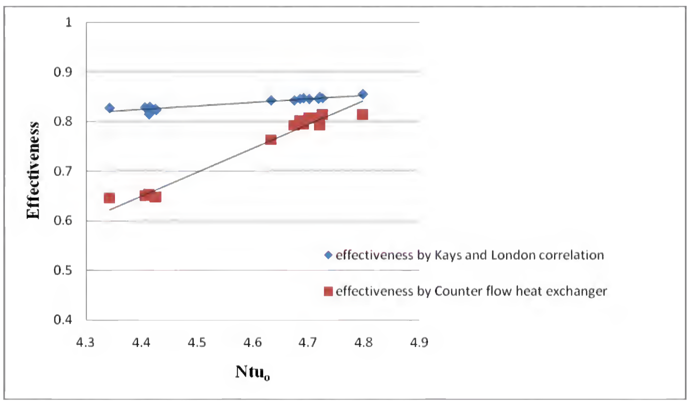

Fig-10: Variation of Effectiveness Vs Ntu_o

The above Fig-9 shows the variation of heat transfer rate (Q) with mass flow rate of air (m_a). The result shows that the heat transfer rate increases with increase in mass flow rate of air due to the rate of heat transfer is directly proportional to the mass flow rate.

The above Fig-8 shows the variation of effectiveness (ε) with Ntu_o. The result shows that the effectiveness by Kays and London correlation increases with increase in number of transfer unit (Ntu_o), similarly the effectiveness of counter flow heat exchanger also increases with Ntu_o. From the figure we noticed that the effectiveness by kays and London is always more than the effectiveness by counter flow heat exchanger. Therefore the performance of air preheater is better than the counter flow heat exchanger.

VI. Conclusions

The main points from the present study can be concluded as follows:

1. Mass flow rate of air varies from 165 to 191.67kg/s and mass flow rate flue gases kept constant at 166.67 kg/s, the thermal performance of the air preheater was analyzed using ε-NTU method and results obtained are Colburn-j factor on air side was decreased by 13.74%, Colburn-j factor on flue gas side was decreased by 2.04%, Friction factor on air side was decreased by 13.45%, Friction factor on flue gas side was decreased by 2%, heat transfer coefficient of air increased by 1%, and pressure drop of air increased by 34.4%.
2. Mass flow rate of air increases from 165 kg/s to 184.21 kg/s. Then the Heat transfer rate is increased by 13.7%.
3. Number of transfer units increases from 4.34 to 4.79. Then the effectiveness by Kays and London correlation is increased by 3.3%.
4. Heat capacity rate ratio increases from 0.9 to 0.99, then the effectiveness of the air preheater by Kays and London correlation decreased by 5%.

NOMENCLATURE

A	overall heat transfer surface area
A_{fr}	rotor cross sectional area, (m^2)
C	flow stream heat capacity rate, (W/K)
C^*	heat capacity rate ratio (C_{min}/C_{max})
C_{min}	minimum of C_h and C_c, (W/K)
C_{max}	maximum of C_h and C_c, (W/K)
C_r	total heat capacity rate of a matrix, (W/K)
C_r^*	total matrix heat capacity rate ratio (C_r/ C_{min})
C_p	specific heat at constant pressure, (J/kg-K)
C_m	specific heat of matrix, (J/kg-K)
D	rotor diameter or disk diameter, (m)
d	hub diameter, (m)
D_h	hydraulic diameter, (m)
F	Friction factor
G	mass velocity, (kg/m^2-s)
h	convective heat transfer coefficient, (W/m^2-K)
$(hA)^*$	symmetry factor related to thermal resistance (hA) on the C_{min} side/ (hA) on the C_{max} side)
J	Colburn j factor for heat transfer (St $Pr^{2/3}$)
L	disk height, (m)
m	mass flow rate, (kg/s)
M	mass of matrix bed, (kg)
N	rotational speed, (rpm)
Ntu_o	number of transfer units
P	pressure, (Pa)
Pr	Prandtl number ($\mu C_p/k$)
Q	rate of heat transfer (kW)
Re	Reynolds number (GD_h/μ)
T	temperature, (^oC)
U_o	overall heat transfer coefficient, (W/m^2-K)

Greek symbols

M	dynamic viscosity, (N-s/m^2)
ε	Effectiveness
ρ	fluid density, (kg/m^3)
ρ_m	matrix density, (kg/m^3)
σ	porosity
Δp	pressure drop, (Pa)

Subscripts

a	air
f	flue gases
o	outlet
I	inlet
max	maximum
min	minimum

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